

1998

Modeling of Wobble Plate Compressors Used in Automotive Air-Conditioning

R. Dirlea
Valeo Climate Control

L. Gauthy
University of Liege

M. Grodent
University of Liege

Y. Khamsi
Valeo Climate Control

J. Lebrun
University of Liege

See next page for additional authors

Follow this and additional works at: <https://docs.lib.purdue.edu/icec>

Dirlea, R.; Gauthy, L.; Grodent, M.; Khamsi, Y.; Lebrun, J.; and Negoiu, D., "Modeling of Wobble Plate Compressors Used in Automotive Air-Conditioning" (1998). *International Compressor Engineering Conference*. Paper 1249.
<https://docs.lib.purdue.edu/icec/1249>

This document has been made available through Purdue e-Pubs, a service of the Purdue University Libraries. Please contact epubs@purdue.edu for additional information.

Complete proceedings may be acquired in print and on CD-ROM directly from the Ray W. Herrick Laboratories at <https://engineering.purdue.edu/Herrick/Events/orderlit.html>

Authors

R. Dirlea, L. Gauthy, M. Grodent, Y. Khamisi, J. Lebrun, and D. Negoiu

MODELING OF WOBBLE PLATE COMPRESSORS USED IN AUTOMOTIVE AIR-CONDITIONING

R. Dirlea⁽¹⁾, L. Gauthy⁽²⁾, M. Grodent⁽²⁾, Y. Khamsi⁽³⁾, J. Lebrun⁽²⁾ and D. Negoiu⁽²⁾

Valeo climate control (⁽¹⁾USA, ⁽³⁾France)

⁽²⁾ Laboratory of Thermodynamics, University of Liège, Belgium

ABSTRACT

The modeling of a wobble plate compressor requires a lot of experimental data covering the whole operating range. Two test benches are specially designed to meet this purpose; they are briefly described in this paper. A special attention is paid to the oil fraction circulating inside the system and a method based on energy balances is proposed to identify this oil fraction. The compressor model is based on a physical approach. Despite the very wide operating range, it is shown that good predictions of both gas mass flow rate and shaft power can be obtained using a limited number of parameters.

INTRODUCTION

The compressor model describes in this paper is a module of a software package developed for the design and simulation of automotive air-conditioning systems. The wobble plate compressor is chosen as an example in order to illustrate the modeling method; however, other types of compressors (such as the rotary vane compressor and the scroll compressor) are also modeled as part of this software package.

OBJECTIVES OF THE COMPRESSOR MODELS AND LITERATURE SURVEY

The main objectives of a compressor model are to predict the refrigerating capacity, the shaft/electrical power consumption (or the corresponding COP), the lubricant mass flow rate and the heat exchange with environment (or the corresponding refrigerant temperature at compressor exhaust). Most of traditional models only generate the two first outputs: lubricant flow rate and environmental heat exchange are generally neglected. In most traditional applications, the lubricant flow rate does not matter because of two main reasons: or it is actually negligible (for example, with low speed scroll compressors), or a separator is available at compressor exhaust (for example, with big screw compressors : high lubricant flow rate is required in order to reduce internal leakage, to cool the refrigerant and to reduce mechanical frictions). However, in many applications, environmental heat exchanges should not be neglected, specially for small and low speed compressors. Car air-conditioning is performed with compact compressors operating over a large domain of rotation speed. The lubricant flow rate is therefore never negligible. On the other hand, environmental heat exchange may have to be taken into account for low speed regime. At higher speeds, it appears that the lowering of refrigerant exhaust temperature is mainly due to the lubricant circulation (the effect of the environmental heat exchange is then of secondary importance). Examples of new models dealing with refrigerant flow rate and shaft power predictions are presented in this paper. New models of lubricant flow rate and heat exchange are still under development.

Black box models are widely used in refrigeration. They consist in correlating refrigerating capacity and shaft/electrical power to evaporating and condensing temperatures with some reference assumptions (and/or correction laws) about subcooling (at condenser exhaust) and superheating (at evaporator exhaust). The experimental correlations are usually expressed in a polynomial form, very easy to use in the simulation. The main drawback of this approach is its high experimental cost: a large amount of experimental data is required in order to identify all the coefficients of each compressor over its whole operating range. This is due to the fact that the coefficients do not have any physical meaning and therefore cannot be transposed in any way from one compressor to another one.

Some of the previous models are based on simple evaluations of volumetric and isentropic effectiveness. These characteristics are sometimes assumed to remain constant within a limited domain of use, or to be related to one selected variable as the pressure ratio.

Some authors proposed to keep a polynomial approach in order to define the refrigerating capacity, but to replace the power characteristic by a COP estimate. This COP is sometimes defined through a “Carnot” efficiency (ratio of the actual compressor COP to the COP of a Carnot cycle with the same reference temperatures) [1]. Other attempts were made to substitute Rankine to Carnot as reference cycle, but without any benefit in accuracy [2].

Efforts should not have to be spent on the choice of a reference cycle but rather invested in a more physical analysis of the compressor behavior. The ASHRAE Toolkit model [3] is oriented towards this physical approach. For stationary constant speed reciprocating compressors, the refrigerant flow rate can be accurately defined by means of a correlation using the outside pressure ratio as the only independent variable. Two parameters have to be identified for this flow rate model: the swept volume and the clearance factor (ratio of the clearance volume to the swept volume). Both parameters have fictitious values: the difference from actual values comes from neglecting physical phenomena other than re-expansion of clearance volume (among others, the effect of supply pressure drop). Another linear model has been proposed to predict the shaft/electrical power. Again two parameters are to be identified: a constant electromechanical loss and a proportional loss factor applied to the isentropic compression power.

This very simple model proved to be fairly accurate most of the time, but not always: some compressors can diverge from this law at very low pressure ratios. An additional parameter was then introduced in the model: a fictitious exhaust throttling area.

The main advantages of using a physically meaningful model are that it “consumes” less parameters (and therefore less experimental data) and that it allows to get default data. These default data will provide the user with at least first guesses when a new compressor has to be simulated.

The ASHRAE Toolkit model has been validated on the basis of a systematic compilation of many manufacturer catalogues and also through many experiments. A similar physical approach is used in this study in order to model the wobble plate compressor. This model is based on new laboratory test results.

DESCRIPTION OF THE TEST BENCH

Two test benches were designed in order to cover a large experimental domain and to allow a systematic data acquisition usable for further development of a compressor model.

The following specifications were to be met:

- One test bench should allow compressor tests with rotation speeds between 1000 rpm and 3000 rpm and the other one between 3000 rpm and 5000 rpm;
- Refrigerant to be used: HFC-134a;
- Variation of the evaporating pressure: 2 bar (29 psia) to 5 bar (72.5 psia);
- Variation of the condensing pressure: 10 bar (145 psia) to 30 bar (435.1 psia);
- Variation of the superheating at compressor supply: 0 K (0 F) to 20 K (36 F);
- The compressor should be kept in an ambience of 40 C (104 F);
- The wobble plate compressor should operate between 20% and 100% of its maximum displacement.

The refrigerant circuit of each test bench includes a water-cooled condenser, a downstream pressure controller used as expansion valve and a water-heated evaporator. The compressor is installed inside a thermostatic calorimeter. The measurement of the mixture (refrigerant and oil) mass flow rate is obtained by means of a Coriolis type flowmeter located at the condenser exhaust. The shaft power transmitted to the compressor is determined from rotation speed and torque measurements (the torque is measured using a dynamometric method).

MASS AND ENERGY BALANCES

One of the major difficulties of the experimental study lies in the high oil fraction contained in the total flow rate (the oil fraction, κ_{oil} , is defined as the ratio of the oil mass flow rate to the total mass flow rate): oil fractions up to 15 % are observed. An indirect method has been developed in order to identify the oil fraction. This method is based on a combined energy balance of the three main components: the compressor, the evaporator and the condenser. The residual of each component energy balance is defined as follows:

$$\dot{R} = \left(\frac{dU}{dt} \right) - \dot{M}_{\text{mix}} (h_{\text{mix},\text{su}} - h_{\text{mix},\text{ex}}) - \dot{M}_w c_w (t_{w,\text{su}} - t_{w,\text{ex}}) - \dot{Q}_{\text{amb}} - \dot{W}_{\text{sh}} \quad (1)$$

where $\left(\frac{dU}{dt} \right)$ is the change of internal energy (negligible in steady-state conditions) [W];

\dot{M}_{mix} is the mixture (refrigerant and oil) mass flow rate [kg s^{-1}];

h_{mix} is the mixture specific enthalpy [J kg^{-1}];

\dot{M}_w is the water mass flow rate [kg s^{-1}];

t_w is the water temperature [C];

c_w is the water specific heat [$\text{J kg}^{-1} \text{K}^{-1}$];

\dot{Q}_{amb} is the heat flux supplied to the component by its environment [W];

\dot{W}_{sh} is the shaft power [W];

Subscripts “su” and “ex” stand for “supply” and “exhaust” conditions, respectively;

$\dot{M}_w (h_{w,\text{su}} - h_{w,\text{ex}}) = 0$ for the compressor and $\dot{W}_{\text{sh}} = 0$ for the evaporator and the condenser.

The specific enthalpy of oil-refrigerant mixture is a function of the oil fraction, of the solubility (i.e. the refrigerant concentration in liquid phase) and of the specific enthalpies of each component (refrigerant – liquid phase and vapor phase – and oil). The solubility of refrigerant in oil (ζ) depends on the nature of the refrigerant, on the nature of the oil and on the pressure-temperature conditions. An empirical correlation for a mixture of HFC-134a/polyalkylene glycol (PAG) [4] is used in this work.

The following merit function is built for each test using the relative values of the three residuals:

$$\Phi = \left(\frac{\dot{R}_{\text{cp}}}{\dot{W}_{\text{sh}}} \right)^2 + \left(\frac{\dot{R}_{\text{ev}}}{\dot{M}_{w,\text{ev}} c_w (t_{w,\text{su}} - t_{w,\text{ex}})_{\text{ev}}} \right)^2 + \left(\frac{\dot{R}_{\text{cd}}}{\dot{M}_{w,\text{cd}} c_w (t_{w,\text{ex}} - t_{w,\text{su}})_{\text{cd}}} \right)^2 \quad (2)$$

This function is minimized considering the three following cases:

- (i) The working fluid is pure refrigerant: the flow rate is identified from energy balances (one unknown: \dot{M}_{ref});
- (ii) The working fluid is a mixture of refrigerant and oil: the total flow rate is obtained from the Coriolis mass flowmeter while the oil fraction is identified from energy balances (one unknown: κ_{oil});
- (iii) The working fluid is a mixture of refrigerant and oil: both total flow rate and oil fraction are identified from energy balances (two unknowns: \dot{M}_{mix} and κ_{oil}).

The results of these calculations show that neglecting the oil circulation leads to very important errors (the relative residual of the compressor energy balance is about 5 % for $\kappa_{\text{oil}} = 6$ % and goes up to 25 % for $\kappa_{\text{oil}} = 16$ %). On the other hand, a very small residual (lower than 5 % over the whole variation range of κ_{oil}) is obtained when the oil circulation is considered (calculation with or without the Coriolis flowmeter indication). The oil fraction subsequently used is identified using the Coriolis flowmeter indication. There is a quite satisfactory agreement between this value and the oil fraction obtained by a sampling method: the difference between the two methods lies generally around 1 % (in absolute value).

MODELING OF THE WOBBLE PLATE COMPRESSOR

A wobble plate compressor (or variable displacement compressor) automatically changes its swept volume (by adjusting the piston stroke) to match the system cooling requirement. This concept provides a continuously operating compressor with the two following advantages as compared to a system that cycles the compressor on and off to control the capacity:

- (1) Smooth continuous compressor operation (no cyclic load applied to the engine);

(2) Protection of the evaporator against icing.

The control characteristic can be represented by a relationship between the suction and discharge pressures: $p_{su, cp} = f(p_{ex, cp})$.

Since the working fluid is a mixture of refrigerant (vapor and liquid phases) and oil (liquid phase), the model of the actual compressor can be split into two fictitious subsystems: a gas compressor in parallel with a liquid pump. It is clear that the compression work developed by the pump is negligible as compared to the one developed by the compressor. The model must therefore lead to an accurate prediction of the gas mass flow rate (refrigerant in vapor phase) passing through the compressor. The gas mass flow rate at compressor supply can be conventionally selected as an output variable of the model $\left(\dot{M}_{g, su} = \left(\frac{1 - \zeta_{su} - \kappa_{oil}}{1 - \zeta_{su}} \right) \dot{M}_{max} \right)$:

$$\dot{M}_{g, su} = f(v_{su}^*, p_{su}^*, p_{ex}, N) \quad (3)$$

where v_{su}^* and p_{su}^* are the refrigerant specific volume [$m^3 kg^{-1}$] and pressure [Pa] before the compression, p_{ex} is the pressure after the compression and N is the rotation speed [Hz].

Equation (3) is an approximation: it neglects the effects of some “secondary” variables such as lubricant charge and surrounding temperature as well as the effects of some refrigerant properties such as its viscosity.

The first step of the modeling is to introduce a throttling pressure drop at compressor supply. This throttling is fictitiously decomposed into two steps. The fluid is first assumed to be expanded through a fictitious nozzle; this expansion is adiabatic and reversible (therefore also isentropic). It is followed by an adiabatic and fully irreversible (therefore also isobaric) diffusion. This throttling is globally isenthalpic. If the pressure drop is “not too large” (i.e. much smaller than the entrance pressure), the fluid may be considered as quasi-incompressible and then it remains:

$$p_{su} - p_{su}^* = \frac{\dot{M}_{g, su} v_{su}}{2 \left(\frac{\pi d_{su}^2}{4} \right)^2} \quad (4)$$

where d_{su} is the diameter of the fictitious nozzle throat area [m].

The compressor volumetric effectiveness is mainly affected by the re-expansion of the fluid trapped in the clearance volume. This effect is very well identified for reciprocating compressors and it can be described by the following equation if an isentropic compression process is assumed:

$$\frac{\dot{M}_{g, su} v_{su}^*}{N} = V_s - V_c \left(\frac{v_{su}^*}{v_{ex, s}^*} - 1 \right) \quad (5)$$

where $v_{ex, s}^*$ is the specific volume associated with pressure p_{ex} and specific entropy s_{su}^* [$J kg^{-1} K^{-1}$]; V_s is the swept volume [m^3] and V_c is the clearance volume [m^3].

The flow rate characteristic is thus defined by means of three parameters: d_{su} , V_s and V_c . It is obvious that the identification can only be carried out for the tests associated with a maximum displacement operation (for the tests with a reduced displacement operation, the actual displacement is unknown). A very interesting result is that the identified swept volume differs from the actual value (provided by the manufacturer) only by a few per cent. This confirms the physical meaning of the model. The actual swept volume can be used in place of the corresponding parameter. This very good “default” information makes that only two parameters are still to be identified for the refrigerant flow rate model. An example of such an identification is presented in figure 1; it is based on a series of 55 tests with maximum displacement. The gas mass flow rate prediction varies between about -6 % and +8 %.

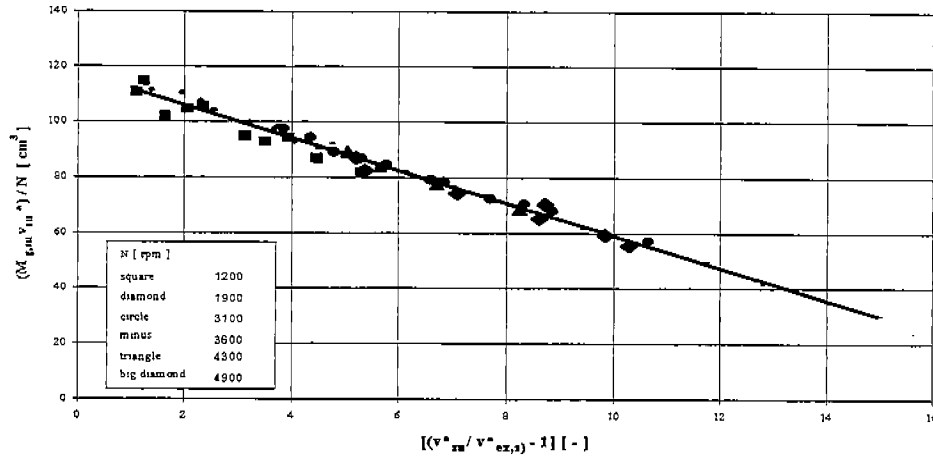


Figure 1 - Flow rate characteristic

Equations (4) and (5) can be used in order to determine the displacement ratio (DR: ratio of the actual swept volume to the maximum swept volume) for the tests associated with a reduced displacement. Indeed, the reduced swept volume can be calculated in order to find the experimental gas mass flow rate, keeping constant the values identified for the clearance volume and for the equivalent diameter.

The compressor shaft power can be split into several terms:

- An isentropic compression power: $\dot{W}_s = \dot{M}_{g,su} (h_{ex,s}^* - h_{su})$;
- Mechanical losses proportional to the isentropic compression power: $\alpha(N) \dot{W}_s$, where $\alpha(N) = C_\alpha \left(\frac{N}{N_0} \right)^m$ (N_0 is a reference rotation speed: $N_0 = 16.67$ Hz or 1000 rpm). This formulation comes from the observed linear dependence of the shaft power with the isentropic power for a given rotation speed;
- Mechanical losses associated to a constant torque (for a given displacement ratio): $C_T N$ [Nm].

When the compressor operates with a reduced displacement, the mechanical losses due to a constant torque should be smaller. Indeed, we can consider that this constant torque is produced by the friction between the pistons and the cylinders. The compressor reduces its swept volume by reducing its stroke which consequently leads to a reduction of the resistive torque. The torque C_T can be (fictitiously) split into two components:

$$C_T = C_{T0} + C_{T1} \cdot DR$$

where C_{T0} is the friction torque at zero displacement ratio;

$(C_{T1} \cdot DR)$ is the friction torque proportional to the displacement ratio.

The shaft power model is obtained by combining all these terms:

$$\dot{W}_{sh} = \dot{W}_s + C_\alpha \left(\frac{N}{N_0} \right)^m \dot{W}_s + (C_{T0} + C_{T1} \cdot DR) N \quad (6)$$

where C_α , m , C_{T0} and C_{T1} are positive parameters to be identified.

The result of this identification is shown in figure 2. As can be seen, a very satisfactory correlation is obtained around the bisecting line (a series of 80 tests with both maximum and reduced displacements are considered). The error is globally lower than ± 10 %; the highest relative errors are encountered for low shaft powers (i.e. relatively small errors in absolute value).

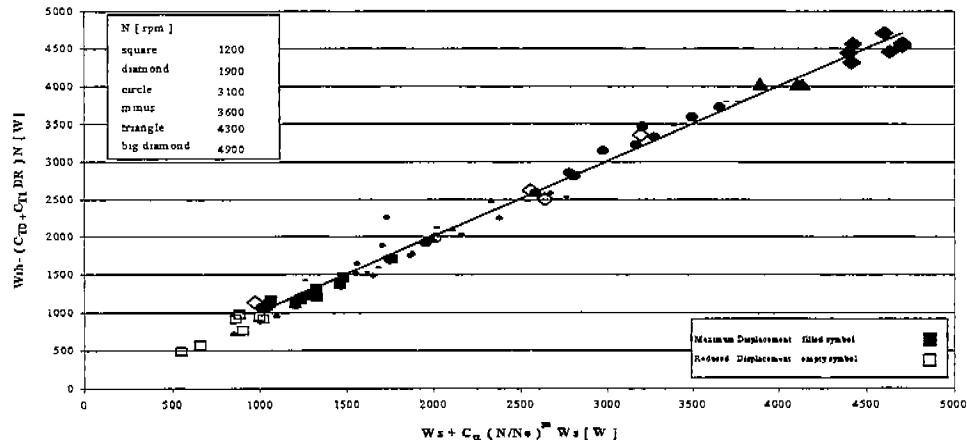


Figure 2 - Shaft power characteristic

The results presented here are obtained for a single compressor. The model has also been successfully validated on the basis of tests carried out with two other wobble plate compressors. In each case, the clearance factor

$\left(\frac{V_c}{V_{s,max}} \right)$ remains approximately the same (about 5 %); the same conclusion is also valid for the dimensionless

ratio $\frac{d_{su}}{\sqrt[3]{V_{s,max}}}$ (about 0.14). Default data can also be proposed for the shaft power characteristic: $C_\alpha=0.16$;

$m=0.58$; $\frac{C_{T0}}{V_{s,max}} = 50000 \text{ Pa (7.25 psia)}$; $\frac{C_{T1}}{V_{s,max}} = 100000 \text{ Pa (14.5 psia)}$ (these two last values refer to “equivalent”

pressures as for combustion engines). So it is possible to recommend default data if a new compressor of the same technology is to be simulated without any preliminary test results available.

CONCLUSIONS

The wobble plate compressor is modeled using a physical approach. Only two parameters are required to predict the gas mass flow rate: the clearance volume and the diameter of the fictitious nozzle throat area located at compressor supply. It is assumed that the shaft power is split into three main components: the isentropic compression power, the mechanical losses proportional to the isentropic compression power (and depending on the rotation speed) and the mechanical losses associated to a friction torque (depending on the displacement ratio). Four parameters must be identified. The error on the shaft power prediction is globally lower than $\pm 10 \%$ over the whole operating range.

REFERENCES

- [1] H.C. Peitsman and H.J. Nicolaas. 1988. Liquid chilling system. TNO Institute of Applied Physics, Delft, The Netherlands. Report prepared for the International Energy Agency, Annex 10: System Simulation.
- [2] O.S. Hernandez, J. Lebrun, C. Saavedra, C.A. Silva and K.L. Silva. 1990. Chiller and heat pump models: towards a better taking into account of compressor characteristics. System Simulation in Buildings, Proceedings of the third international conference, Liège, Belgium.
- [3] J.P. Bourdouxhe, M. Grodent and J. Lebrun. 1996. HVAC1KIT – A Toolkit for primary HVAC system energy calculation prepared for the American Society of Heating, Refrigerating and Air-Conditioning Engineers (TC 4.7 Energy Calculation). Laboratory of Thermodynamics, University of Liège, Belgium.
- [4] J.J. Grebner and R.R. Crawford. 1993. Measurement of pressure-temperature concentration relations for mixtures of R-12/mineral oil and R-134a/synthetic oil. ASHRAE Transactions, Vol. 99, Part 1, USA.